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ABSTRACT

Experiments and analyses were performed to determine ball bearing cooling requirements at shaft speeds to 35 000 rpm and thrust loads to 400 pounds (1780 N). A bronze-filled polytetrafluoroethylene (PTFE) retainer was used to lubricate the bearings. Bearing outer-race temperature varied as the inverse of hydrogen gas flow rate through the bearing. From this inverse relation, a minimum flow equation was developed for the range of bearing operating conditions. The minimum flow rate required for bearing cooling was more sensitive to speed than to load increases. Film transfer to the inner-race grooves and retainer wear are indicated and discussed.

STAR Category 15

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SUMMARY

The cooling requirements for 40-millimeter-bore ball bearings operating in hydrogen gas at 60° R (33° K) were investigated at shaft speeds of 20 000, 30 000, and 35 000 rpm and at thrust loads of 200, 300, and 400 pounds (890, 1335, and 1780 N). A bronze-filled polytetrafluoroethylene (PTFE) composition was used as the retainer material. The hydrogen coolant gas flow rates through the bearings ranged from 0.0060 to 0.045 pound per second (0.0027 to 0.0204 kg/sec).

An equation was developed from heat-transfer analysis and experimental data that established the minimum flow rate of hydrogen coolant gas required by the bearings for a range of operating conditions. The investigation showed that minimum flow rates at all test conditions were more sensitive to increases in speed than to increases in thrust load. The experimental results showed that bearing outer-race temperature varied as the inverse of the coolant-gas flow rate through the bearing.

Postrun inspection of the bearings indicated that the bronze-filled PTFE retainers provided adequate transfer films on the load carrying surfaces and prevented inner-race wear. Bearing retainer wear was light in all bearings, varying from 0.85 to 2.76 weight percent.

INTRODUCTION

Ball bearings used in rocket-engine turbopumps for pumping low-density liquid hydrogen operate at high rotative speeds and moderate thrust loads for short periods. The bearing loads are moderate because the large pressure force developed by the pump is normally equalized by a balance-pressure device built into the turbopump. A maximum degree of reliability is required by the turbopump for several minutes running time (total run time including checkout is several hours). To ensure high reliability, the bearings must be properly lubricated and adequately cooled.

Lubrication of the bearings is accomplished using retainers fabricated from self-lubricating polytetrafluoroethylene (PTFE) compounds (refs. 1 to 6). The transfer mechanism by which the bearing load carrying surfaces are lubricated with retainer material is described in reference 1 as follows: "As the bearing rotates, the balls rub against the retainer. Thin films of retainer material are transferred to the balls and subsequently to the race grooves. The retainer locating surface on one of the races is lubricated directly by sliding contact with the retainer material."

Bearing cooling techniques have been developed for liquid-hydrogen turbopumps. Cooling methods have progressed from immersion studies at ambient pressure in a liquid-hydrogen bath (ref. 2), to high pressure, high flow of liquid hydrogen through the bearings (ref. 3). Adequate bearing cooling can also be provided by using hydrogen gas at cryogenic temperature (ref. 1). It is anticipated however that, at high rotative speeds, cooling with gaseous hydrogen subjects the bearings to more severe conditions than cooling with liquid hydrogen. These conditions result in higher retainer wear and lower cooling capacity. To achieve maximum efficiency of the rocket engine, it is most desirable that the turbopump bearings operate with the minimum flow of hydrogen required for cooling regardless of the cooling mode used.

The objectives of this investigation were (1) to determine experimentally the relation between the hydrogen-coolant-gas flow rate and the bearing outer-race temperature for a range of bearing speeds and thrust loads, and (2) to develop an equation for minimum coolant-gas flow rate based on the experimental results.

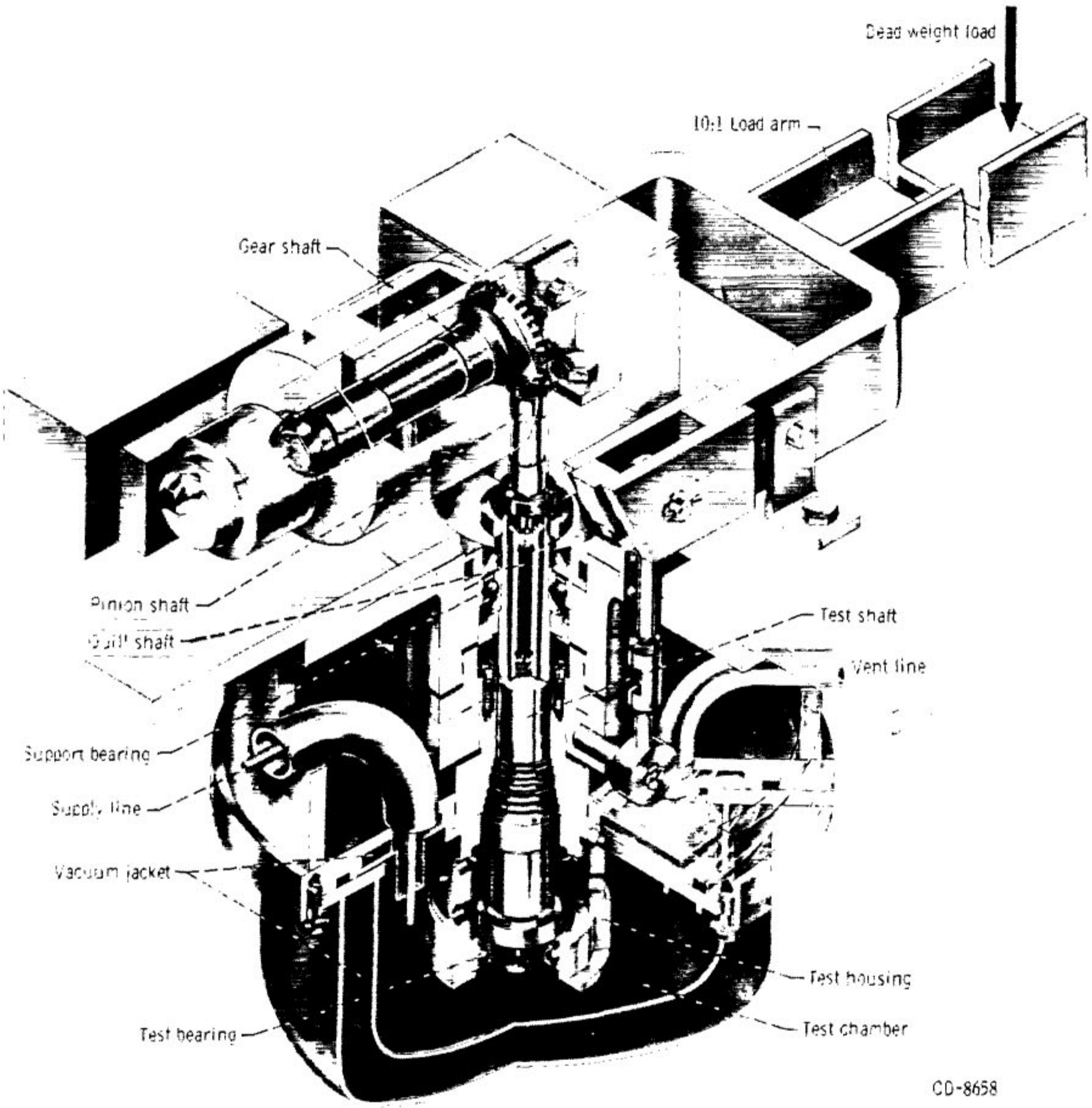
The experimental study was conducted with 40-millimeter-bore ball bearings operating in hydrogen gas at a constant inlet temperature of 60° R (33° K), at coolant-gas flow rates from 0.006 to 0.045 pound per second (0.0027 to 0.0204 kg/sec), at shaft speeds from 20 000 to 40 000 rpm, and at thrust loads of 200, 300, and 400 pounds (890, 1335, and 1780 N). The retainer was a 30-weight-percent bronze-filled PTFE. The selection of this material was based on its film-transfer capabilities and low wear for running times up to 10 hours in gaseous hydrogen at 60° R (33° K) (ref. 1).

An analysis, based on the heat transferred from the bearing through convection to the hydrogen-coolant gas, was made to establish a relation between bearing temperature and hydrogen flow rate. This relation, in conjunction with experimental data, was used to determine the minimum required flow rates through the bearing for the range of operating conditions investigated.

APPARATUS

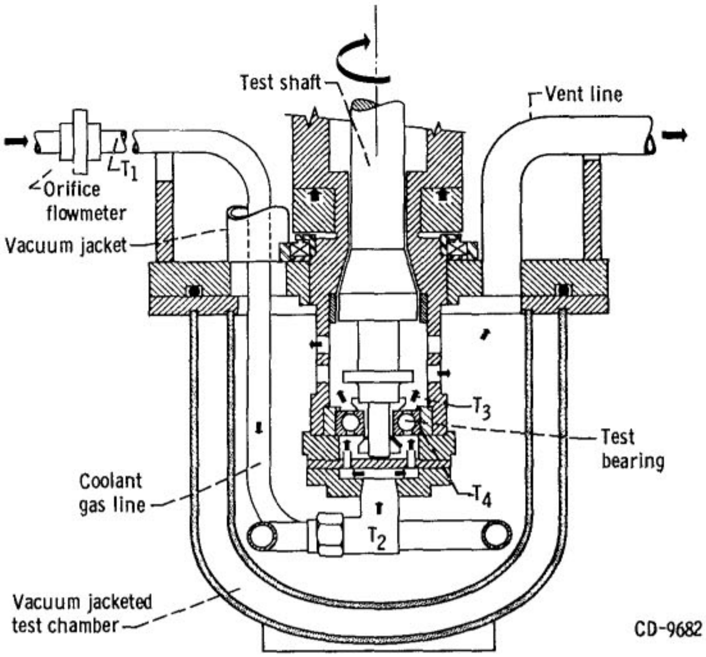
Bearing Test Rig

The test apparatus shown in figure 1 was first described in reference 2. The test bearing is driven through a gear assembly by a variable-speed, direct-current motor.



(a) Load system and test-shaft mounting.

Figure 1. - Cryogenic fuel bearing test rig (ref. 2).



(b) Test-bearing mounting and coolant-gas supply.
Figure 1. - Concluded.

Automatic speed control (to within ±0.1 percent) can be provided over a range of test-shaft speeds from 900 to 52 500 rpm. The test shaft was supported at its lower end by the test bearing and at its upper end by an oil-lubricated ball bearing. Thrust load was applied to the test-bearing housing from a deadweight load (fig. 1(a)). A schematic of the test-bearing mounting and support housing is shown in figure 1(b). Temperature symbols shown thereon are defined in appendix A.

Hydrogen Supply and Exhaust System

The test bearing was cooled by a direct stream of cryogenic hydrogen gas (fig. 1(b)). A schematic of the bearing coolant flow system is shown in figure 2. Liquid hydrogen from the Dewar vaporized in a counterflow shell-and-tube heat exchanger. Heat was supplied by flowing hydrogen gas at ambient temperature through the shell. The cryogenic hydrogen-gas flow rate to the test bearing was measured by an orifice flowmeter located in the coolant-gas supply line downstream from the heat exchanger. The range of the coolant-gas flow rate was from 0.006 to 0.045 pound per second (0.0027 to 0.0204 kg/sec). After flowing through the test bearing, the coolant gas was exhausted from the test chamber through the vent line. Both the liquid-hydrogen flow from the Dewar and

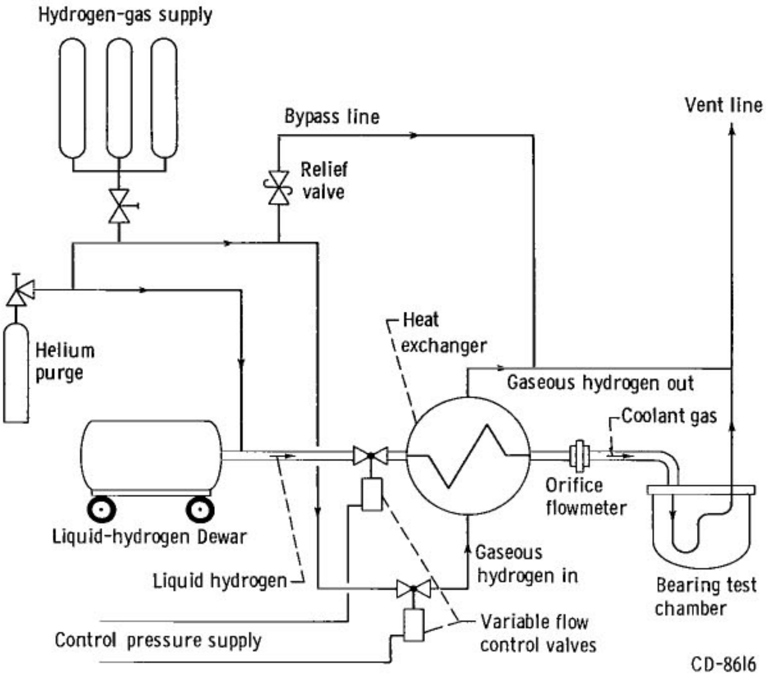


Figure 2. - Schematic of test-bearing coolant flow system.

the warm hydrogen-gas flow to the heat exchanger were regulated by remote-control variable-flow valves. The locations of these valves are shown in figure 2. The test chamber, coolant-gas supply line, and liquid-hydrogen line were vacuum-jacketed. The hydrogen gas line from the heat exchanger was insulated with 1/2-inch (1.27-cm) thick polyurethane.

Temperature Measurement

Platinum-resistance sensors were used to measure cryogenic temperatures at the following locations (fig. 1(b)): coolant-gas supply line downstream from the orifice flowmeter (T_1), (2) hydrogen-gas inlet to the test bearing (T_2), (3) hydrogen-gas exhaust downstream of the test bearing (T_3), and (4) bearing outer race (T_4). Temperatures could be read to an accuracy of $\pm 0.3^{\circ}$ R ($\pm 0.17^{\circ}$ K) at 60° R (33° K) with the data recording system.

Copper-constantan thermocouples were used to measure cryogenic temperature in the vent line. Additional copper-constantan thermocouples were located in the support-bearing housing and in the gearbox housing in the region of the high-speed shaft.

TABLE I. - SUMMARY OF MINIMUM FLOW RESULTS

[Deep-groove ball bearings, 40-mm bore, separable at outer race; race and ball material, 440C stainless steel; number of balls, 10; ball diameter, 0.375 in. (0.953 cm); inner-race curvature, 0.54; outer race curvature, 0.58; radial clearance 0.0025 in. (0.0064 cm); retainer material, 30 percent bronze-filled PTFE; retainer inner land clearance, 0.038 in. (0.097 cm); ball pocket clearance 0.015 in. (0.038 cm); inlet gas temperature, 60° R (33° K).]

Bearing	Thrust load		Speed, rpm	Constant, C_1		Constant, C_2		Correlation coefficient	Minimum coolant flow rate[a]		Outer-race temperature at minimum coolant flow rate	
	lb	N		°R	°K	(lb)(°R)/sec	(kg)(°K)/sec		lb/sec	kg/sec	°R	°K
8B	200	890	20 000	58.8	32.7	0.0492	0.0124	0.945	0.0035	0.0016	73	41
			30 000	60.6	33.7	.2053	.0517	.993	.0072	.0033	89	49
			35 000	62.1	34.5	.5289	.1333	.993	.0115	.0052	108	60
15B	300	1335	20 000	59.9	33.3	0.0617	0.0155	0.972	0.0039	0.0018	76	42
			30 000	57.4	31.9	.2988	.0753	.973	.0086	.0039	92	51
			35 000	49.9	27.7	.6099	.1537	.943	.0124	.0056	98	54
16B	400	1780	20 000	58.9	32.7	0.0922	0.0232	0.990	0.0048	0.0022	78	43
			30 000	53.8	29.9	.3827	.0964	.999	.0098	.0044	93	52
			35 000	33.8	18.8	.9780	.2464	.971	.0157	.0071	93	52
21B	200	890	30 000	59.8	33.2	0.1115	0.0281	0.976	0.0053	0.0024	81	45
	300	1335	30 000	56.2	31.2	.2285	.0576	.970	.0076	.0034	86	48
	400	1780	30 000	56.2	31.2	.2548	.0642	.998	.0080	.0036	88	49

[a]See eq. (C3).

Test Bearings and Retainers

The bearings used in these tests were 40-millimeter bore (108 series) deep-groove ball bearings manufactured to ABEC-5 tolerances. One shoulder on the outer race was relieved to make the bearings separable. The inner- and outer-race curvatures were 0.54 and 0.58, respectively. The average radial clearance was 0.0025 inch (0.0064 cm). The ball and race material were AISI 440C stainless steel.

The retainers were of one-piece machined construction without reinforcement. They were located on the inner-race with an average land clearance of 0.038 inch (0.097 cm) and an average ball pocket clearance of 0.015 inch (0.038 cm) (table I).

PROCEDURE

Pretest Procedure

The bearings were prepared for testing in the following manner: they were (1) degreased with trichloroethylene, (2) inspected and measured for clearances, (3) checked with a surface measuring instrument to obtain surface profiles of the inner-race grooves, (4) washed in trichloroethylene, (5) stored in vacuum-desiccator chamber for approximately 6 hours, (6) removed from the desiccator, separated into components and the individual components weighed prior to testing, (7) reassembled, and (8) stored in a desiccator until used.

Test Procedure

After the bearing was installed in the test housing, the test chamber and all hydrogen lines were purged for 15 minutes with helium gas. After the purge operation, cold hydrogen gas was force-fed to the bearing. The test shaft was rotated at 900 rpm during cool-down. The thrust load was applied early in the 10-minute cool-down period. When the system reached the temperature equilibrium, the shaft speed was increased to the test speed (in 5000-rpm increments every 5 min).

The flow rate of hydrogen gas was initially set at a high value, approximately 0.040 pound per second (0.018 kg/sec). This flow was maintained until temperature equilibrium was achieved in the system and until the first data point was taken, which required a total of 30 minutes. Temperature equilibrium was assumed for each subsequent data point when all the temperatures had remained constant for a minimum of 3 minutes prior to recording the data. Total time at temperature equilibrium, including data recording time, was normally 5 minutes. Constant shaft speed, thrust load, and inlet temperature were maintained for each data point. The four cryogenic temperatures (described in the Temperature Measurement Section, p. 5) were recorded continuously throughout each test.

The range of hydrogen coolant-gas flow rates was from 0.006 to 0.045 pound per second (0.0027 to 0.0204 kg/sec). Prior to any change in bearing speed, a data point previously obtained at a high flow rate was checked. Bearing outer-race temperature was normally repeatable to within 2° R (1.1° K) at these high flow rates.

A flow map was obtained for a bearing at a constant thrust load of 200 pounds (890 N) at shaft speeds of 20 000, 30 000, and 35 000 rpm by varying the flow rate as described previously. This procedure was repeated with other bearings, using the same retainer material, at 300- and 400-pound (1335- and 1780-N) thrust loads. A flow map was also obtained for a bearing at a constant speed of 30 000 rpm for thrust loads of 200, 300, and 400 pounds (890, 1335, and 1780 N).

The coolant-gas-supply pressure, measured at the inlet to the test bearing, ranged from 14.6 to 15.6 pounds per square inch absolute (10.0 to 10.8 N/cm^2 abs). The 500-gallon (1.89-m^3) supply of liquid hydrogen provided enough coolant gas for a run-time of approximately 120 minutes after the cool-down period and setting of the first flow point.

Postrun Inspection of Bearings

After each test, the system was purged with helium gas. The test bearing was (1) inspected for wear, (2) washed in trichloroethylene, and (3) placed in a vacuum desiccator for approximately 6 hours.

At the end of a run series, each bearing was weighed to determine the weight loss or gain (in mg) of each component. Transverse surface profile traces were made on the inner-race grooves, in the region of the pretest surface profiles, to determine the extent of race wear and/or film buildup from transferred retainer material. The balls, races, and retainers were examined visually and with optical microscopy to help determine the extent of wear and surface damage. Photographs of the retainers were made to illustrate the wear patterns that occurred during the test period.

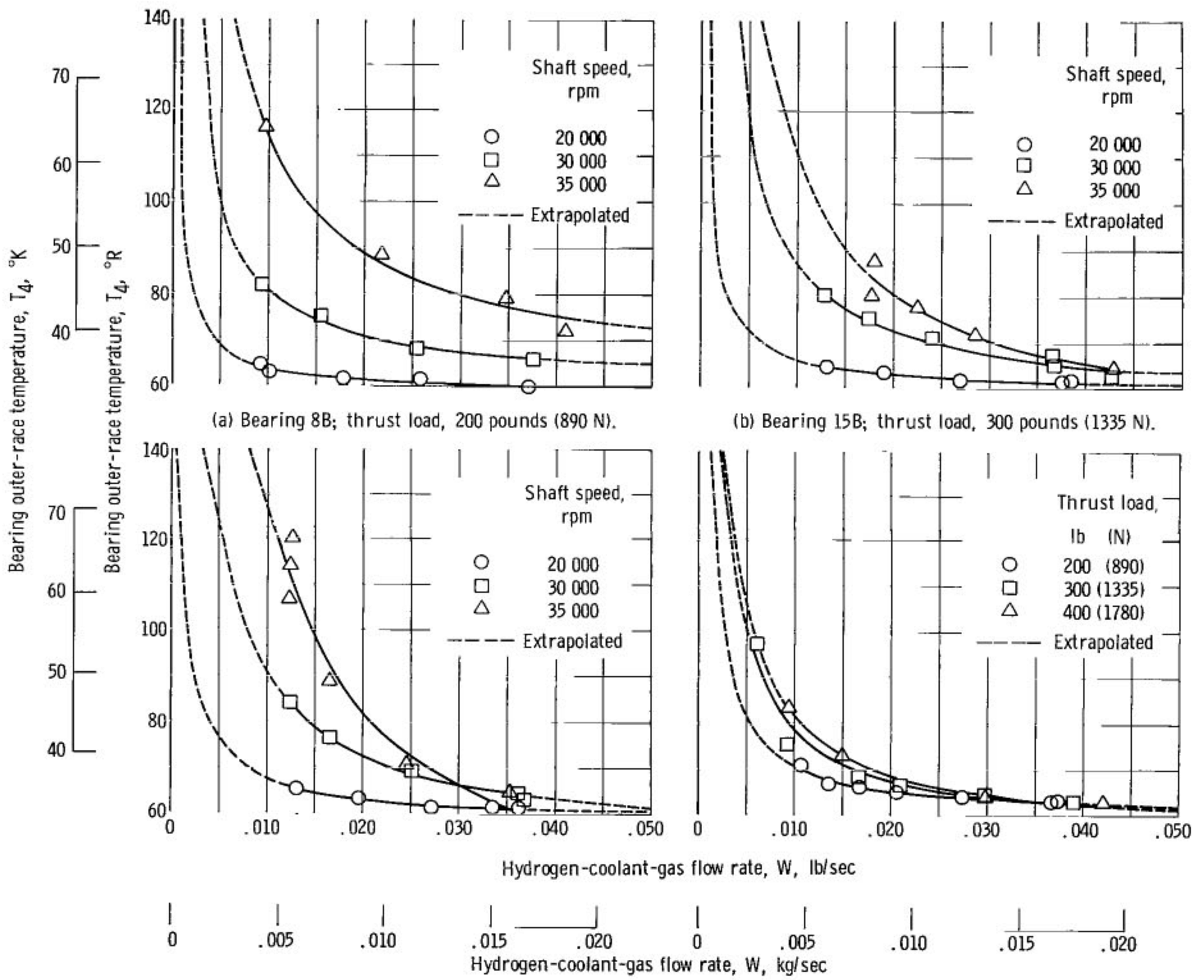
RESULTS AND DISCUSSION

Experimental Test Results

<u>Effect of flow rate on bearing outer-race temperature</u>. - Ball bearings with bronze-

filled PTFE retainers were run at constant speed and load conditions while flowing hydrogen gas at 60° R (33° K) through the bearings. These experimental results are shown in figure 3. The effect of coolant flow rate on bearing outer-race temperature was determined for speeds of 20 000, 30 000, and 35 000 rpm at loads of 200, 300, and 400 pounds (890, 1335, and 1780 N) for bearings 8B, 15B, and 16B, respectively, (figs. 3(a) to (c)). Additionally, the effect of flow rate on outer-race temperature of bearing 21B at a constant speed of 30 000 rpm at loads of 200, 300, and 400 pounds (890, 1335, and 1780 N) is shown in figure 3(d). The trend of the data indicates that, as the coolant flow rate through the bearing is decreased, the bearing outer-race temperature increases.

In order to determine the functional relation between the bearing outer-race temperature and the flow rate of hydrogen-coolant gas through the bearing, a heat-transfer



(a) Bearing 8B; thrust load, 200 pounds (890 N).

(b) Bearing 15B; thrust load, 300 pounds (1335 N).

(c) Bearing 16B; thrust load, 400 pounds (1780 N).

(d) Bearing 21B; shaft speed, 30 000 rpm.

Figure 3. - Bearing outer-race temperature as function of hydrogen-coolant-gas flow rate. Inlet-gas temperature, 60° R (33° K).

analysis was made by considering the heat transferred from the bearing to the coolant gas by convection (see appendix B). The analysis indicated that bearing outer-race temperature should vary as the inverse of the flow rate. Using the relation $T_4 = C_1 + C_2/W$ (eq. (B8)), the curves of figure 3 were determined by a least-squares fit with the data. The curves were extrapolated for flow rates lower than those indicated by the experimental points and are shown as the dashed portion of the curves.

The correlation coefficients, in addition to the constants (C_1 and C_2) for the curves of figure 3, are given in table I for all runs. The correlation coefficients are defined as the ratio of the standard deviation of the experimental data from the mean to the standard deviation of the curve-fit data from the mean. An indication that the equation $T_4 = C_1 + C_2/W$ provided a good representation of the relation between bearing outer-race temperature and coolant-gas flow rate within the range of the experimental data was that all curves showed correlation coefficients of 0.943 or better.

Sensitivity of bearing temperature at low flow rates. - The experimental points shown in figure 3 represent bearing outer-race temperatures measured at the corresponding flow rates. At these flow rates, the bearings remained at the indicated equilibrium temperatures for a minimum of 3 minutes.

The outer-race temperature of bearing 8B, running at 35 000 rpm with a 200-pound (890-N) load, was initially 116° R (65° K) at a flow rate of 0.0097 pound per second (0.0045 kg/sec) (fig. 3(a)). However, after remaining at this value for approximately 4 minutes, the temperature increased to over 150° R (83° K) within an additional 30 seconds. The speed had to be decreased and the coolant flow rate increased to prevent complete bearing failure. The equation $T_4 = C_1 + C_2/W$ predicts that, at low flow rates, bearing temperature is very sensitive to flow change. A small decrease in flow could therefore result in a sharp rise in bearing temperature. Such a rise in temperature could cause the bearing to lose clearance, thus leading to accelerated heat generation in the bearing and eventual seizure.

An attempt to determine minimum flow rate of bearing 8B at 40 000 rpm and 200-pound (890-N) load was unsuccessful because the temperature equilibrium at the bearing outer race could not be established over the range of coolant-gas flow investigated.

Minimum Coolant Flow Rate

Definition of minimum flow. - Minimum coolant flow rate can be defined as the lowest hydrogen gas flow that should be used in operating a bearing at a given speed and load condition. For this investigation, the criterion selected was the sensitivity of the bearing outer-race temperature to changes in coolant flow rate. Thus, the minimum flow rate is more specifically defined as that flow rate at which the bearing temperature sensitivity becomes critical (i.e., the temperature increase for a corresponding flow decrease becomes larger than desirable). For this investigation, a critical sensitivity of 4000° R per pound per second (4890° K/(kg/sec)) was selected, based on the slope of the 35 000-rpm curve for bearing 8B (fig. 3(a)).

The minimum flow rate was therefore calculated from the following equation (derived in appendix C):

$$W_{min} = \sqrt{\frac{C_2}{\left|\frac{dT_4}{dW}\right|}}$$

As noted previously, the values of C_2 were determined for each speed and load condition from the least-squares analysis and $|dT_4/dW|$ is the critical sensitivity (4000° R/(lb/sec) or 4890° K/(kg/sec)). The results are tabulated in table I. It should be noted that most of these calculated minimum flow rates are lower than the range of the experimental data.

Effect of bearing speed and thrust load on minimum flow rate. - In order that the minimum flow equation (C3) apply for any bearing operating conditions in the range investigated, the constant C_2 must be expressed in terms of bearing speed N and thrust load P. The effect of bearing speed on minimum flow rate was determined by plotting C_2 as a function of speed at constant load on suitable log-log coordinate paper. The slope of the line represented the power of the speed dependence. The result indicated that the minimum required gaseous-hydrogen flow rate for bearings using bronze-filled PTFE retainers varies as the square of the speed. Minimum flow as a function of bearing speed is shown in figure 4 for the three loads investigated.

The effect of thrust load on minimum flow was determined in a similar manner by plotting C_2 as a function of load at constant speed. These results indicated that the minimum flow rate varies as the square root of the applied load. Minimum flow rate as a function of thrust load is shown in figure 5 for speeds of 20 000, 30 000, and 35 000 rpm. In order to verify the effect of thrust load on minimum flow rate, without including the data scatter from using different bearings, bearing 21B was run at a constant speed of 30 000 rpm at loads of 200, 300, and 400 pounds (890, 1335, and 1780 N). The calculated values of minimum flow rate for bearing 21B (shown in table I) are also plotted on figure 5. Although these minimum flow rates are lower than those for the derived curve at 30 000 rpm, they do indicate the same general trend with load.

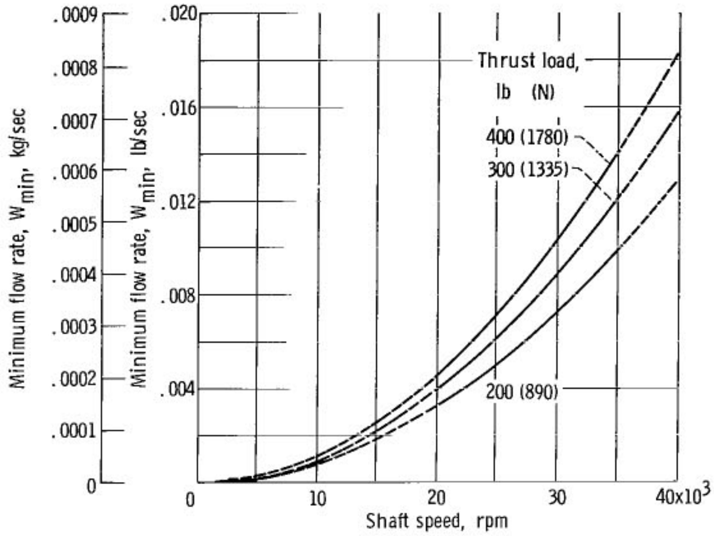


Figure 4. - Minimum hydrogen-coolant-gas flow rate as function of shaft speed for several values of thrust load. Inlet gas temperature, 60° R (33° K).

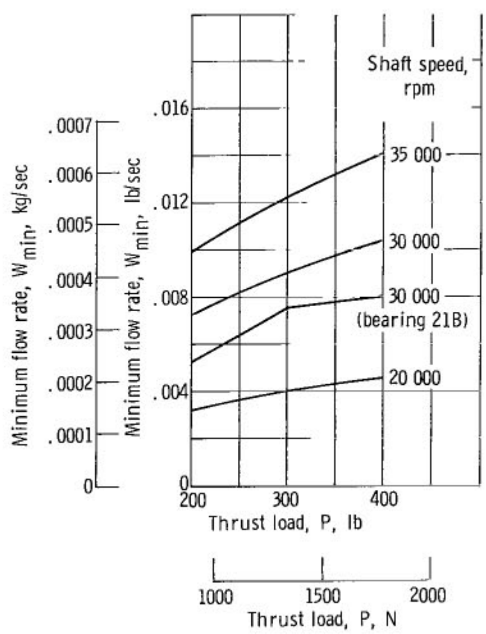


Figure 5. - Minimum hydrogen-coolant-gas flow rate as function of thrust load for several values of shaft speed. Inlet gas temperature, 60° R (33° K).

The final equation for minimum coolant flow rate (with speed and load effects included) for 40-millimeter ball bearings cooled by 60° R (33° K) hydrogen gas for the range of bearing operating conditions investigated can be written as

$$\left.\begin{aligned} W_{min} &= 0.575\times10^{-12}\, N^2\sqrt{P} \qquad \text{(lb/sec)} \\ &\text{or} \\ W_{min} &= 0.118\times10^{-12}\, N^2\sqrt{P} \qquad \text{(kg/sec)} \end{aligned}\right\} \tag{C5}$$

If sharp increases in outer-race temperature are to be avoided, the flow rates of hydrogen-coolant gas through the bearing should be greater than the minimum flows indicated by equation (C5) or shown in figures 4 and 5 for the range of speeds and thrust loads investigated. Comparison of the curves of figures 4 and 5 indicate that the minimum flow rates are more sensitive to changes in speed than in load.

Postrun Examination

At the end of the run series, the components of each bearing were examined with a microscope at a magnification of 15. The extent of film transfer or surface damage on the balls and races and the degree of retainer wear were observed. The run times, retainer wear, sliding distance of the retainer at the inner race, and apparent postrun condition of the bearings are given in table II.

Transfer film observations. - Postrun profile traces were made on the inner-race grooves to indicate the thickness of retainer film transferred during the bearing run series. The profile traces in the ball-track region are shown in figure 6 for bearings 15B, 16B, and 21B. The film buildup on 15B measured more than 150 microinches (4 μm) on either side of the ball running track (fig. 6(a)). Visual examination showed a bronze-colored film nonuniformly distributed across the ball track. This nonuniform film was continuous around the inner-race periphery for approximately 270°. For the remaining 90° the film was thin with uniform distribution. A similar nonuniform film buildup was observed on the inner race of bearing 8B. The bronze film was wider, however, than on bearing 15B, extending over the centerline of the race groove. At speeds above 35 000 rpm, bearing 8B had experienced several high-temperature excursions. Because of the high temperatures, the bearing apparently lost clearance as indicated by the position of the ball track on the inner-race groove.

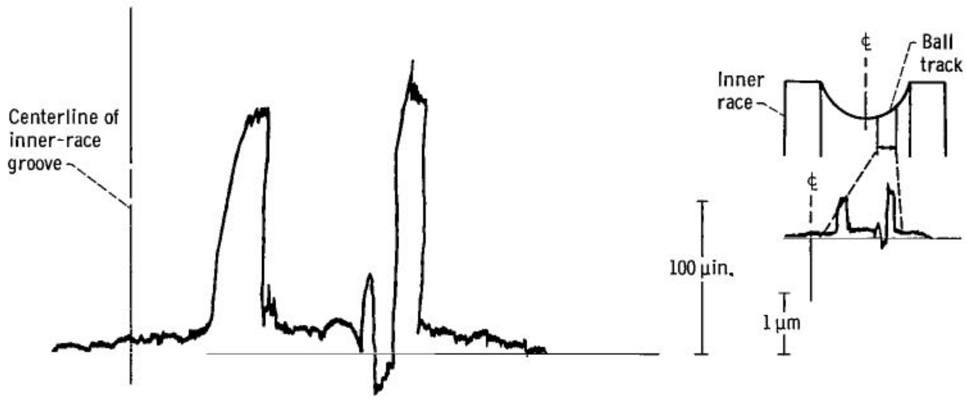
The thickness of the films transferred to the inner-race grooves of bearings 16B and 21B are shown in figures 6(b) and (c), respectively. The films measured approximately

TABLE II. - POSTRUN BEARING CONDITION

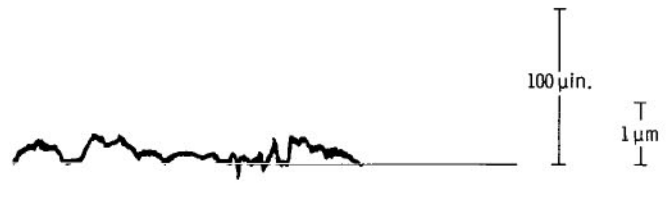
[Coolant flow rate, 0.006 to 0.045 lb/sec (0.003 to 0.021 kg/sec); hydrogen gas temperature, 60° R (33° K).]

Bearing	Shaft speed, rpm	Load		Run time, min		Retainer				Postrun condition		
		lb	N	At each test condition	Total[a]	Total wear, wt. %	Sliding distance at inner race		Wear rate, wt. % per million sliding feet (0.3048×10^{6} m)	Retainer	Inner race	Balls
							ft	m				
8B	20 000	200	890	105	436	1.72	3.34×10^{6}	1.02×10^{6}	0.52	High wear on inner diameter; low pocket wear (fig. 7(b))	Wide film of varying thickness across race groove	Two parallel wear tracks near center (fig. 7(b))
	30 000	↓	↓	120	↓	↓	↓	↓	↓			
	35 000			63								
	40 000			80								
15B	20 000	300	1335	76	305	2.32	2.26×10^{6}	0.69×10^{6}	1.03	High wear for 180° of inner diameter; low pocket wear	Wide film of varying thickness across race groove (fig. 6(a))	Even thickness film
	30 000	300	1335	79	305	2.32	2.26	.69	1.03			
	35 000	300	1335	101	305	2.32	2.26	.69	1.03			
16B	20 000	400	1780	112	350	0.85	2.06×10^{6}	0.62×10^{6}	0.41	High wear for 180° of inner diameter; extremely low pocket wear (fig. 7(a))	Even thickness film (fig. 6(b))	Wide wear track near center (fig. 7(a))
	30 000	400	1780	63	350	.85	2.06	.62	.41			
	35 000	400	1780	59	350	.85	2.06	.62	.41			
21B	30 000	200	890	118	652	2.76	4.94×10^{6}	1.51×10^{6}	0.56	High wear for 180° of inner diameter; medium pocket wear	Even thickness film (fig. 6(b))	Even thickness film
		300	1335	246	652	2.76	4.94	1.51	.56			
		400	1780	170	652	2.76	4.94	1.51	.56			

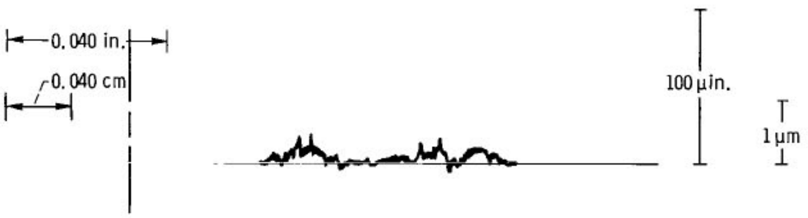
[a]Including run time during cool-down.



(a) Bearing 15B; run time, 305 minutes; shaft speed, 20 000, 30 000, and 35 000 rpm; thrust load, 300 pounds (1335 N).



(b) Bearing 16B; run time, 350 minutes; shaft speed, 20 000, 30 000, and 35 000 rpm; thrust load, 400 pounds (1778 N).



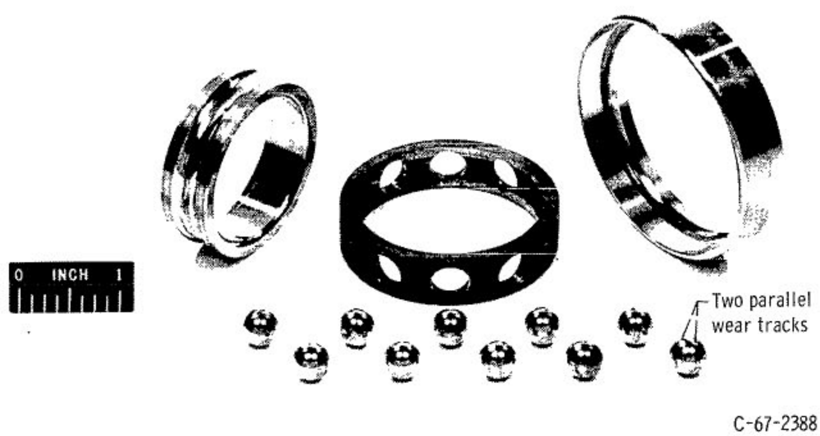
(c) Bearing 21B; run time, 652 minutes; shaft speed, 30 000 rpm; thrust load, 200, 300, and 400 pounds (890, 1335, and 1778 N).

Figure 6. - Profile traces of inner-race ball track (normal to ball-rolling direction).

20 microinches (0.5 μm) and were continuous across the ball track for both bearings. The film transfer characteristics of these two bearings are consistent with those reported in reference 1 for bearings using the same retainer material. In reference 1, bearings run at 20 000 rpm and 200-pound (890-N) thrust load had similar thin, continuous, bronze-colored, films on the inner-race grooves.

The bronze films on the outer-race grooves of all the bearings appeared uniformly thick and evenly distributed around the periphery. The balls of bearings 8B, 15B, and 21B were covered with bronze film, whereas those of bearing 16B showed only a wide bronze wear track near the ball centerline (fig. 7(a)). In addition to the bronze-colored

(a) Bearing 16B. Shaft speed, 20 000, 30 000, and 35 000 rpm; thrust load, 400 pounds (1780 N); run time, 350 minutes; retainer wear, 0.85 percent by weight.



(b) Bearing 8B. Shaft speed, 20 000, 30 000, 35 000, and 40 000 rpm; thrust load, 200 pounds (890 N); run time, 436 minutes; retainer wear, 1.72 percent by weight.

Figure 7. - Bearing retainer wear. Coolant, hydrogen gas at 60° R (33° K).

film covering the surface, two distinct wear tracks were apparent on the balls of bearing 8B (fig. 7(b)). The wear track near the ball center probably resulted from the run time at 20 000 rpm, whereas the other wear track occurred during the 40 000 rpm run. A computer analysis using the bearing internal geometry specified (table I) showed that at a 200-pound (890-N) thrust load, the contact angle divergence (β_i - β_o, the difference between the inner-race (β_i) and outer race (β_o) contact angles) is greater at 40 000 than at 20 000 rpm. This fact indicates that the balls will run at two different wear tracks at the two rotative speeds.

Retainer wear. - The retainer wear values in table II are given as weight percent loss for the total run times shown. The wear of the four bearing retainers was predominantly at the inner-land locating surface (table II and fig. 7). Retainer wear was light in all bearings, varying from 0.85 to 2.76 weight percent.

When retainer wear is compared on the basis of weight percent per million sliding feet (table II), bearing 15B shows the highest wear rate, although differences in wear rates were not significant. Because the retainer wear measurements were made at the end of the bearing run series and not after each bearing run, any observations made as to whether retainer sliding velocity (bearing 8B) or thrust load (bearing 21B) have the greater influence on retainer wear appear to be inconclusive for the bearings run in this investigation.

SUMMARY OF RESULTS

The cooling requirements of 40-millimeter-bore ball bearings operating in hydrogen gas at 60° R (33° K) were investigated for shaft speeds to 35 000 rpm at thrust loads to 400 pounds (1780 N). A bronze-filled PTFE composition was used as the retainer material. The hydrogen-coolant-gas flow rates through the bearings ranged from 0.0060 to 0.0450 pound per second (0.0027 to 0.0204 kg/sec). The minimum required coolant flow of hydrogen gas through the bearing for a given speed and load condition was based on the sensitivity of the bearing outer-race temperature to changes in coolant-gas flow rate. The study produced the following results:

1. Experimental results showed that bearing outer-race temperature varied as the inverse of the hydrogen-coolant-gas flow rate through the bearing.

2. From the experimental results an equation was developed that can be used to determine the minimum flow rates of coolant gas required by the bearings in the range of operating conditions investigated. Minimum flow requirements for the bearings were more sensitive to increases in speed than to increases in thrust load.

3. Evidence of the formation and buildup of transfer films on the inner race from the retainer material was provided by profile traces of the bearing inner-race ball track.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, January 30, 1968,
129-03-13-01-22.

APPENDIX A

SYMBOLS

A	effective heat-transfer area, ft^2; m^2
C_p	specific heat at constant pressure, Btu/(lb)(^oR); J/(kg)(^oK)
C_1, C_2	constants
h	convective heat-transfer coefficient, Btu/(sec)(ft^2)(^oR); W/(m^2)(^oK)
K	constant in minimum flow equation
K_1, K_2	constants
k	proportionality constant between T_4 and T_s
N	shaft speed, rpm
P	bearing thrust load, lb; N
Q_{brg}	heat generated by bearing, Btu/sec; W
Q_{gas}	heat transferred to coolant gas, Btu/sec; W
T_s	temperature of simulated heat-transfer surface, ^oR; ^oK
T_1	temperature of coolant gas downstream from orifice flowmeter, ^oR; ^oK
T_2	temperature of coolant gas at inlet to bearing, ^oR; ^oK
T_3	temperature of coolant gas at outlet from bearing, ^oR; ^oK
T_4	temperature of bearing outer race, ^oR; ^oK
ΔT_{av}	average temperature difference between coolant gas and simulated surface, ^oR; ^oK
$\Delta T_{log\ mean}$	log mean temperature difference between coolant gas and simulated surface, ^oR; ^oK
W	coolant-gas flow rate, lb/sec; kg/sec
W_{min}	minimum flow rate of coolant gas, lb/sec; kg/sec

APPENDIX B

DETERMINATION OF RELATION BETWEEN BEARING OUTER-RACE TEMPERATURE AND COOLANT-GAS FLOW RATE

In order to determine the cooling requirements for a specific operating condition, the relation between the bearing outer-race temperature T_4 and the flow rate W of hydrogen coolant gas through the bearing must first be established. A simplified heat-transfer analysis was made to determine the functional relation of T_4 and W. In the analysis, the heat transferred from the bearing to the coolant gas is simulated by convective heat transfer from a perfectly insulated surface (see fig. 8). The approximate relation was derived using the following simplifying assumptions:

(1) The heat transferred from the bearing Q_{brg} is constant at a constant shaft speed N and a constant thrust load P.

(2) The inlet temperature T_2 of the gas is constant.

(3) The convective heat-transfer coefficient h remains constant.

(4) The specific heat C_p is constant.

(5) The temperature of the surface T_s remains constant over the entire surface.

(6) The bearing outer-race temperature T_4 varies in the same manner as the temperature of the surface T_s.

Furthermore, if it is assumed that all heat generated at the (bearing) surface is transferred by convection to the gas, a heat balance equation can be written as

$$Q_{brg} = Q_{gas} \tag{B1}$$

where

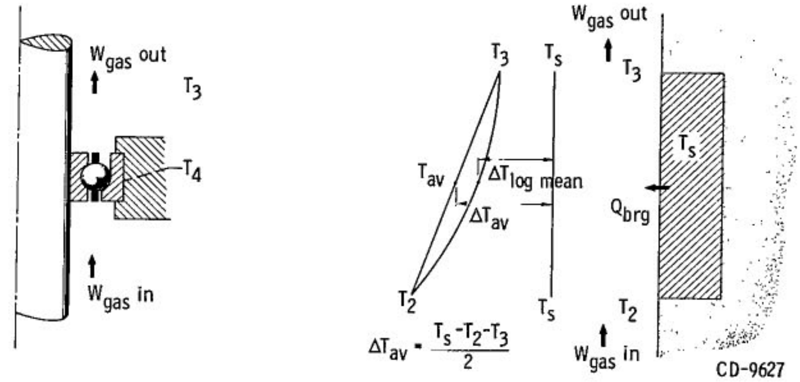
$$Q_{brg} = hA\ \Delta T_{log\ mean} \tag{B2}$$

and

$$Q_{gas} = WC_p(T_3 - T_2) \tag{B3}$$

If it assumed that $(T_3 - T_2)$ is small, then ΔT_{av} is a sufficient approximation to $\Delta T_{log\ mean}$ (fig. 8). From figure 8,

$$\Delta T_{av} = T_s - \frac{T_2 + T_3}{2} = \frac{2T_s - T_2 - T_3}{2} \tag{B4}$$



(a) Bearing cross-section.

(b) Simulated convective heat-transfer surface.

Figure 8. - Coolant-gas flow through bearing.

therefore,

$$Q_{brg} = hA\left(\frac{2T_s - T_2 - T_3}{2}\right) \tag{B5}$$

Solving equation (B5) for T_s and substituting the value of T_3 from equation (B3) as

$$T_3 = \frac{Q_{gas}}{WC_p} + T_2$$

yields

$$T_s = \frac{Q_{brg}}{hA} + T_2 + \frac{Q_{gas}}{2WC_p} \tag{B6}$$

From equation (B1) and using assumptions (1) to (4) then,

$$T_s = \text{Constant} + \frac{\text{Constant}}{W}$$

or

$$T_s = K_1 + \frac{K_2}{W} \tag{B7}$$

From assumption (6),

$$T_4 = kT_s$$

then

$$T_4 = C_1 + \frac{C_2}{W} \tag{B8}$$

Equation (B8) expresses the relation between bearing outer-race temperature T_4 and hydrogen-gas flow rate for a constant speed and load condition. The constants C_1 and C_2 were determined for each set of experimental data by a least-squares analysis and are given in table I.

APPENDIX C

DEVELOPMENT OF MINIMUM FLOW EQUATION

The minimum coolant flow rate has been defined in the RESULTS AND DISCUSSION section as that point at which the bearing temperature (measured at the outer race) becomes too sensitive to changes in coolant flow rate. In other words, at flow rates less than minimum, the bearing outer-race temperature increase for a corresponding flow decrease becomes larger than desirable. Also, the value for the maximum rate of temperature change was selected as 4000° R per pound per second. This rate is actually the slope dT_4/dW. Taking the derivative of equation (B8) with respect to W,

$$\frac{dT_4}{dW} = 0 - \frac{C_2}{W^2} \tag{C1}$$

or

$$W^2 = \frac{-C_2}{\frac{dT_4}{dW}}$$

and, because the slope dT_4/dW is negative,

$$W^2 = \frac{C_2}{\left|\frac{dT_4}{dW}\right|} \tag{C2}$$

Therefore, for minimum flow,

$$W_{min} = \sqrt{\frac{C_2}{\left|\frac{dT_4}{dW}\right|_{max}}} \tag{C3}$$

This equation was used to compute the values of minimum flow rate in this report. The values of C_2 were obtained from least-squares analysis and are given in table I.

In order to obtain a more generalized correlation for the minimum flow, the dependence of C_2 on bearing speed N and thrust load P must be established. This was determined empirically by plotting C_2 against speed at constant load and C_2 against load at constant speed. These plots showed that

$$C_2 \propto PN^4$$

so that equation (C3) becomes

$$W_{min} = KN^2\sqrt{P} \tag{C4}$$

Over the range of bearing test conditions investigated, the value of K was determined as 0.575×10^{-12} lb$^{1/2}$/(sec)(rpm)^2. Thus, the minimum coolant-gas flow rate is related to speed and thrust load as follows:

$$W_{min} = 0.575\times10^{-12} N^2\sqrt{P} \tag{C5}$$

or, for the SI system,

$$\left(W_{min} = 0.118\times10^{-12} N^2\sqrt{P}\right)$$

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